Case Studies in Thermal Engineering 5 (2015) 98–103

Contents lists available at ScienceDirect

Case Studies in Thermal Engineering

journal homepage: www.elsevier.com/locate/csite

Prediction of small spark ignited engine performance using producer gas as fuel

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ARTICLE INFO

Article history:
Received 27 October 2014
Received in revised form
19 January 2015
Accepted 15 February 2015
Available online 16 February 2015

Keywords:
Mathematical modeling
Engine simulation
Small engine
Producer gas
Engine performance

ABSTRACT

Producer gas from biomass gasification is expected to contribute to greater energy mix in the future. Therefore, effect of producer gas on engine performance is of great interest. Evaluation of engine performances can be hard and costly. Ideally, they may be predicted mathematically. This work was to apply mathematical models in evaluating performance of a small producer gas engine. The engine was a spark ignition, single cylinder unit with a CR of 14:1. Simulation was carried out on full load and varying engine speeds. From simulated results, it was found that the simple mathematical model can predict the performance of the gas engine and gave good agreement with experimental results. The differences were within ±7%.

1. Introduction

Producer gas was derived from biomass via gasification with average calorific value of about 5 MJ/Nm^3 [1]. Presently, the use of 100% producer gas in spark ignition (SI) engine was not successful, because producer gas has low energy density, hence, low power output and efficiency [2]. Recently, increasing performance of producer gas engine can be done by increasing compression ratio (CR), changing combustion chamber, mounting gas carburetor and modifying the ignition system [3,4]. Experimental evaluation of a producer gas engine can be costly, complicated and time consuming. Ideally, the engine performance may be predicted using mathematical equations [5]. Establishing mathematical models is of interest. In this work, a single zone cylinder model was used. It can provide quick calculation of optimum conditions. Examination of various engine performance parameters may be achieved [6,7]. The basic assumption of the single zone cylinder model was based on mass balance analysis, regardless of chemical reaction, homogeneous charges, and mixing of gases inside the cylinder [8]. Therefore, the objective of this work was to study the use of mathematical model in small producer gas engine comparing with experimental in term torque, brake power, thermal efficiency and specific fuel consumption.

2. Mathematical modeling

The model was combined with physical based equations for describing phenomena and performance of the small producer gas engine. The details of the mathematical models are as follows:

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http://dx.doi.org/10.1016/j.csite.2015.02.003

2.1. Cylinder pressure

The pressure in cylinder of SI engine can be derived from the first law analysis. The cylinder pressure versus crank angle is shown in Eq. (1) [9].

$$\frac{dP}{d\theta} = \frac{k-1}{V}\frac{dQ}{d\theta} - k\frac{P}{V}\frac{dV}{d\theta} \tag{1}$$

where, P is the pressure inside cylinder, θ is crank angle, k is specific heat ratio, Q is heat releases, V is the cylinder volume and as a function of crank angle, given as

$$V(\theta) = \frac{V_d}{r_c - 1} + \frac{V_d}{2}\left[\frac{l}{a} + 1 - \cos\theta - \left(\left(\frac{l}{a}\right)^2 - \sin^2\theta\right)^{0.5}\right] \tag{2}$$

where, V_d is displacement volume, r_c is compression ratio, l is connecting rod length, a is crank radius.

2.2. Heat input

The total amount of heat input to cylinder versus changes in the crank angle is shown in Eq. 3 [10].

$$\frac{\partial Q}{d\theta} = HV\int_{IVO}^{IVC} m^{\bullet}\, d\theta \frac{df}{d\theta} \tag{3}$$

where, HV is heating value, $m^{\bullet}$ is producer gas flow rate, IVO and IVC are inlet valve open and close positions before and after TDC, $f(\theta)$ is the Wiebe function. Producer gas flow rate through an intake valve was derived empirically from the engine test run between 1100–1900 rpm of engine speed. It is given as

$$m^{\bullet} = 0.00378V_d(0.105N^2 - 0.7922N - 0.0015N^3) \tag{4}$$

where, N is engine speeds and the Wiebe function is used to determine the combustion rate of the fuel, expressed as [11]:

$$f(\theta) = 1 - \exp\left[-5\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^3\right] \tag{5}$$

where, θ is crank angle, θ_0 is start of heat release angle, $\Delta\theta$ is duration of heat release and can be determined from this equation.

$$\Delta\theta = -1.618\left(\frac{N}{1000}\right)^2 + 19.866\left(\frac{N}{1000}\right) + 39.395 \tag{6}$$

2.3. Heat transfer

The heat transfer is necessary for the internal combustion engine to maintain cylinder walls, pistons and piston rings. Normally, the heat transfer in the combustion engine includes conduction, convection and radiation [12]. However, for an SI engine, the primary heat transfer mechanism from the cylinder gases to the wall is convection, with only 5% from radiation [13]. The heat loss to the wall can be determined from the Newtonian convection equation [14] which is given as

$$Q_{loss} = hA(T_g - T_W) \tag{7}$$

where, h is heat transfer coefficient, A is surface area of combustion chamber T_g is gas temperature in cylinder, T_W is cylinder wall temperature. The heat transfer coefficient is instantaneous area average heat transfer coefficient derived from Woschni [15], shown in Eq. (8).

$$h = 0.82b^{-0.2}(P10^{-3}c)0.8T_g^{-0.53} \tag{8}$$

where, b is bore cylinder, c is equal to 6.18. The gas temperature is calculated using following equation from Sitthiracha [16] while, engine speed is in a range of 1000–6000 rpm.

$$T_g = 3.395\left(\frac{N}{1000}\right)^3 - 51.9\left(\frac{N}{1000}\right)^2 + 279.49\left(\frac{N}{1000}\right) + 676.21 \tag{9}$$

Calculation of surface area in cylinder is from the following equation [9] which includes cylinder head, cylinder bore and piston crown. Surface area at any crank angle is given as:

$$A(\theta) = \frac{\pi}{2}b^2 + \pi b\frac{s}{2}\left[\frac{l}{a} + 1 - \cos\theta - \left(\left(\frac{l}{a}\right)^2 - \sin^2\theta\right)^{0.5}\right] \tag{10}$$

2.4. Indicated and brake mean effective pressure

The sums of pressure in cylinder are indicated mean effective pressure (*imep*). The equation is given as [10]:

$$imep = \frac{\oint P\,dV}{V\,d} \tag{11}$$

Therefore, brake mean effective pressure (*bmep*) can be calculated from

$$bmep = imep - \sum fmep \tag{12}$$

2.5. Friction

The friction loss in an internal combustion engine can be analyzed by three components, including the mechanic friction, the pumping work and accessory work. Calculation of engine friction uses an empirical equation [17]. Major frictions include bearing friction, piston and ring friction, wall tension ring friction, valve gear friction, pumping loss, combustion chamber and wall pumping loss. The equations of friction loss are shown in Eqs. (13–18).

$$\text{Bearing friction}\quad fmep_1 = 0.0564\left(\frac{b}{s}\right)\left(\frac{N}{1000}\right) \tag{13}$$

$$\text{Piston and ring friction}\quad fmep_2 = 12.85\left(\frac{P_s}{bs}\right)\left(\frac{100S_l}{1000}\right) \tag{14}$$

$$\text{Wall tension ring friction}\quad fmep_3 = 10\left(\frac{0.377sn_p}{b^2}\right) \tag{15}$$

$$\text{Valve gear friction}\quad fmep_4 = 0.226\left(30 - \frac{4N}{1000}\right)\left(\frac{GD_{iv}}{b^2s}\right) \tag{16}$$

$$\text{Pumping loss}\quad fmep_5 = 0.0275\left(\frac{N}{1000}\right)^{1.5} \tag{17}$$

Combustion chamber and wall pumping loss

$$fmep_6 = 0.0915\sqrt{\frac{imep}{11.45}}\left(\frac{N}{1000}\right)^{1.7} \tag{18}$$

where, P_s is piston skirt length, S_l is mean piston speed, n_p is number of piston ring, G is number of intake valve per cylinder, D_{iv} is Intake valve diameter, P_{mi} is the sum of pressure in cylinder.

2.6. Torque and brake power

The brake power and torque can be determined by following equations:

$$P_b = 0.5bmepNV_d \tag{19}$$

$$T_b = \frac{P_b}{2\pi N} \tag{20}$$

2.7. Brake thermal efficiency and brake specific fuel consumption

The brake thermal efficiency and brake specific fuel consumption (BSFC) when biomass is used as fuel can be modified from gasoline and diesel engine Eqs. [18,19], as

$$\eta_{th} = \frac{P_b}{m^{\bullet} HV} \tag{21}$$

$$BSFC = \frac{m_b^{\bullet}}{P} \tag{22}$$

where, $m_b^{\bullet}$ is biomass (charcoal) consumption

2.8. Initial temperature and pressure of compression process

From the Otto cycle, the first process is isentropic compression. Calculation of initial temperature and pressure can be as follows [17]:

$$\frac{T_2}{T_1} = r_c^{k-1} \tag{23}$$

$$\frac{P_2}{P_1} = r_c^{k} \tag{24}$$

where, T_1 and P_1 are ambient temperature and pressure while T_2, P_2 are cylinder temperature and pressure in the compression process.

3. Experimental setup and measurements

Model validation was carried out through experimentation. A small SI engine converted from a CI engine was used to operate 100% on producer gas. The engine was of single cylinder, four strokes, 598 cc and bathtub combustion chamber [4]. The detailed specifications of small producer gas engine are shown in Table 1. The power output was measured by a dynamometer set and monitored by a display panel. The best experimental conditions were used to develop mathematical models. They were on full load and 14: 1 of CR, the engine speed between 1000–2000 rpm. Producer gas was derived from charcoal. The composition of the gas was of CO 30.5 $\pm$ 2%, H_2 8.5 $\pm$ 2%, CH_4, 0.35%, CO_2 4.8 $\pm$ 1%, and O_2, 6.3 $\pm$ 0:5%, and the balance Nitrogen. The mean calorific value of the producer gas was 4.64 MJ/Nm^3. Parametric study was based on numerical solution to find performance of the engine.

4. Results and discussions

In this study, the small producer gas engine model was developed to estimate torque, brake power, thermal efficiency and specific fuel consumption. The simulated results were compared against the engine experiment. They are shown in Figs. 1–3. At low engine speeds, the predicted values were almost equal to the experimental results. At high speeds, there were small differences at engine speeds between 1500–1900 rpm.

This may be attributed to difference in producer gas flow rate entering the cylinder. The producer gas flow rate was derived empirically from the fuel consumption and volumetric efficiency. However, the deviations were likely due to other factors such as pressure and temperature in cylinder in combustion process. The use of a simple model did not consider

Table 1
Engine and operational specifications in simulation.

Engine type	SI engine, 4 stroke, single cylinder
Fuel	Producer gas
Compression ratio	14:1
Spark ignition timing	30° BTDC
Bore × Stroke (mm)	92 × 90
Connecting rod length (m)	0.143
Crank radius (m)	0.0413
Clearance volume (m^3)	4.60×10^{-5}
Swept volume (m^3)	5.98×10^{-3}
Rated output (kW)	3.2 @ 1700 rpm
Ambient pressure (kPa)	0.92 kPa
Ambien temperature (K)	308
Mean wall temperature (K)	400
Air density (kg/m^3)	1.2
Air/fuel ratio	1.2:1
Equivalent air/fuel ratio	1
Duration of combustion	90°

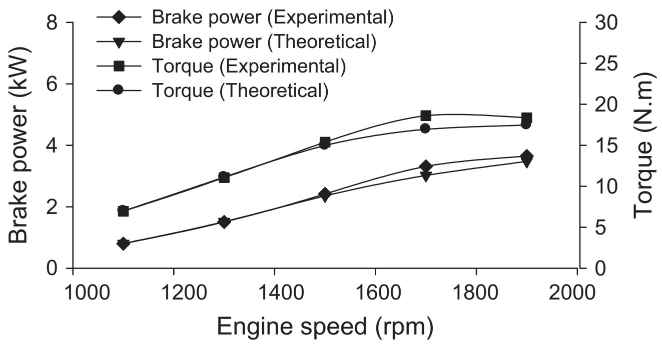

Fig. 1. Comparison between theoretical and experimental brake power and torque.

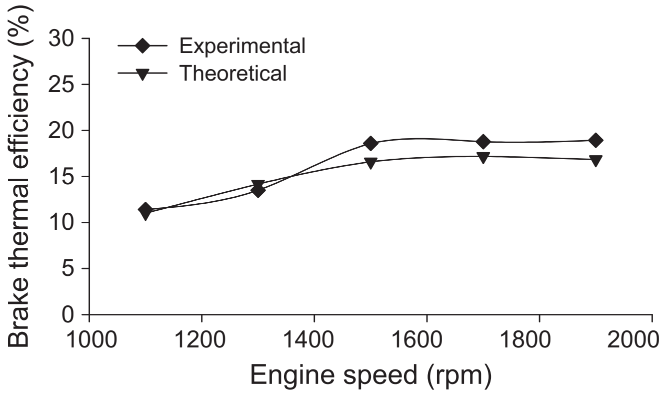

Fig. 2. Comparison between theoretical and experimental brake thermal efficiency.

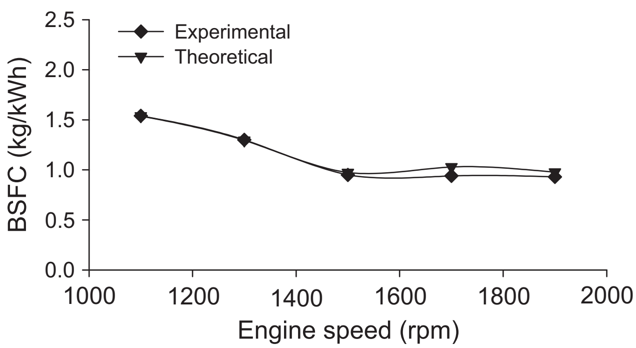

Fig. 3. Comparison between theoretical and experimental brake specific fuel consumption.

micro-analysis of the engine [10]. The average errors of brake power, torque, thermal efficiency and BSFC were −3.30, −3.32, −6.50 and 3.07%, respectively. Therefore, it is concluded that the developed mathematical model gave good agreement and can be applied to the small producer gas engine under the similar conditions.

Table 2
Mean percentage error of thermodynamics model with SI engine.

Engine performance	Mean percentage error (%)			
	This work	[10]	[13]	[20]
Brake power (BP)	−3.30	7.63	−2.74	23.08
Torque	−3.32	–	−3.14	–
Brake thermal efficiency (BTE)	−6.50	0.06	–	21.83
Brake specific fuel consumption (BSFC)	3.07	−0.12	–	–

For comparison, the use of the thermodynamics model to an IC engines is summarized in Table 2. The model validations of the three engines were four stroke SI engine operated on gasoline and gasoline/ ethanol blend. The mean errors of both engines were in a range of -0.12–7.63%. They appeared to be acceptable, compared to the experimental results. The mathematical modeling of this work may be used to predict performance of an SI engine operated on producer gas engine well.

5. Conclusions

The model adopted for this work was found to be acceptable and may be used to predict the performance of producer gas engines. The average percentage errors of brake power, torque, brake thermal efficiency and BSFC were within 6.50%.

Acknowledgment

Supports from Chiang Mai University, the Energy Policy and Planning Office, and the Commission on Higher Education were highly appreciated. The authors would like to thank the Graduate School of Chiang Mai University, as well as the Energy Research Centre of Maejo University for providing test facilities and technical supports.

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